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ANALYSIS OF DASHPOT PERFORMANCE FOR ROTATING CONTROL DRUMS OF A LITHIUM COOLED FAST REACTOR CONCEPT

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January, 1972

ABSTRACT

The reference design of a compact fast reactor for space application uses a rotating control drum concept. A dashpot was incorporated in the design of the drive train of the rotating control drum to prevent shock damage to the control drum and drive train at the termination of a scram action. A rotating vane dashpot using reactor coolant lithium as a damping fluid appears to be the best candidate of the various damping devices explored. This report presents a performance analysis, results and discussion of vane type dashpots for use in this application.

E-6734

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SUMMARY

Damping devices of three different configurations (rotating vanes, bellows, and linear pistons) were considered for use on the rotating control drums of the reference design of a compact fast reactor for space power application. The salient features of these three configurations were compared. The rotating vane seemed to be the best candidate for this application. Therefore performance characteristics were analytically determined for three models of vane type dashpots; two of equal size having different vane shapes (rectangular and circular), and another, a circular vane of smaller size.

The object of the analysis was to determine the braking characteristics produced with different clearances between the rotating vane and the stationary housing and to determine the clearances required to produce uniform deceleration of the control drum.

The analysis showed that to produce the desired braking characteristics, the clearances between the vane and housing are required to taper. The clearances for the three models analyzed were within the range of 0.254 millimeter at the start of braking to 0.0508 millimeter (the minimum clearance permitted by the design clearance of the control drum bearings) at the end.

The analysis also showed that the rectangular vane model is more effective than the circular vane model of the same external size, (i.e., the 14.6 cm diameter x 6.45 cm high space alloted to the dashpot in the reference design). This permits larger and less precise clearances to be used in the rectangular vane design for equal braking performance. The rectangular vane model also appears to be easier to fabricate.

As a consequence of the minimum allowable clearance the final velocity of the control drum is greater than zero when reaching the terminal point. However, this velocity was found to be low enough that the resulting impact would not damage either the control drum or the drive train.

INTRODUCTION

Reactor design studies carried on at Lewis Research Center have shown that moving fuel and poison in and out of the reactor core is an attractive method of controlling a compact fast spectrum reactor. A rotating control drum concept which uses a combination of fuel and neutron absorber in its structure has been evaluated and reported in references 1 and 2. This concept is incorporated in the reference design for a compact fast reactor technology program being conducted at Lewis Research Center for space power applications. Figure 1 is an isometric illustration of this reactor concept.

The reactor core and the six rotating control drums are cooled by liquid lithium at 1220 K. During a reactor scram each control drum is rapidly rotated by a spring operated mechanism (not shown in fig. 1) on the input shaft of the penetration device. The penetration device provides a flexible hermetic seal through which rotary motion is transmitted to the control drum inside the lithium filled pressure vessel.

To prevent possible shock damage to a control drum and drive train during a scram, a dashpot was incorporated in the reference reactor design between the penetration device and the control drum to uniformly decelerate the control drum at the end of the scram movement. A survey was made to find a compact dashpot that could use 1220 K liquid lithium as the damping fluid. The rotating vane dashpot seemed to be the best candidate for this application. A performance analysis was therefore made for three rotating vane geometries. This analysis and its results are described in this report.

DESIGN CONSIDERATIONS

The most severe shock absorbing situation that can arise occurs during a reactor scram near the end of core life, when the control drums must be rapidly rotated π radians. The assumed reference design scram time for this situation is 0.4 second. To control the bearing loads in the drive train within tolerable limits, uniform acceleration in a time period of 0.267 second and uniform deceleration in a time period of a 0.133 second was assumed. With these assumptions the maximum rotational velocity of the control drum is 15.7 radians per second, the rotational displacement during acceleration is 2.1 radians and the rotational displacement during deceleration is 1.05 radians. With the moment of inertia of the control drum being 0.298 kilograms per square meters, the torque imput to the control drum from the scram spring is required to be constant at 17.6 newton meters over the π radians of rotation. Thus during the acceleration time the angular acceleration of the control drum is constant. The dashpot is designed to provide the required uniform deceleration of the control drum.

The dashpots as shown in figure 1 are to be located inside of the pods on the drive shaft housings extending from the lower pressure vessel end cap, between the penetration device and the pressure vessel. The dashpot was placed in this location (near the control drum) rather than at the input end of the shaft, so that the drum will not lose the damping action of the dashpot in the event of a failure of a drive train component during a scram. The damping action for this arrangement would prevent the control drum from bouncing off the stop and coming to rest in an unsafe position. It is also advantageous to place the dashpot in this location from the standpoint of loads on the control drum and drive train bearings.

The dashpot cannot be located inside of the pressure vessel end cap because of interference with the reactor inlet coolant flow distribution. The chosen location which is on the reactor side of the hermetic seal of the penetration device also avoids the problem of providing a moving liquid seal to contain the dashpot fluid.

To avoid having to pay too large a penalty in shield weight, the dashpot pod diameter will not be allowed to exceed the diameter of the penetration device, (approximately 14.6 cm) and the height will be kept to a minimum.

Survey of Damping Devices

All of the dashpot configurations explored work by using the rotational energy of the control drum to force a fluid through an orifice. The types of dashpots looked at were: vane type, bellows type, and linear piston.

I. Vane Type

A rotating rectangular vane dashpot was considered for use as the damping device. An example of this configuration is shown in figure 2. The advantages of this type of dashpot are:

(1) Simplicity; only one moving part.
(2) Required clearances and alignment are such that friction between moving parts is very low.
(3) Very low impact upon striking the stop.
(4) The braking rate can be controlled by varying the clearance.

The main disadvantage of the vane type dashpot is that accurate clearances between the vane and the housing are required in order to obtain the desired deceleration.

Rotating vane dashpots with circular vanes such as the one shown in figure 3 were also explored. The advantages and disadvantages are basically the same as for the rectangular configuration. The one advantage is that the leakage of fluid around the center shaft is eliminated. But the reduced weight flow caused by having a lower vane area than a comparable size rectangular type produces lower braking torques. The circular vane configuration also appears to be the more difficult of the two to fabricate.

II. Bellows Type

An example of an energy absorbing device using a bellows is shown in figure 4. This type of device was considered in an effort to avoid close machining tolerances and the exact alignment required by the rotating vane dashpots. Although these drawbacks are successfully avoided in this type of dashpot, there are several disadvantages:

(1) The maximum length bellows that can be used in this design occupies 4.2 radians of the circular track minus the thickness of the orifice wall. The remaining 2.1 radians of the circular track has to be open to permit the drum to accelerate. The required stroke of the bellows during the braking period is 1.05 radians. Therefore, the stroke of the bellows must be 25 percent of its curved length. Part of the available stroke of the bellows has already been taken up by bending the bellows into the circular shape. As a result the bellows must be stroked to its recommended limit and bellows life becomes a question.

(2) The bellows rubs against the outside wall of the channel as it is stroked.

(3) An impact occurs when the pivot arm strikes the end cap of the bellows at the start of braking.

(4) The orifice size is constant throughout the braking period, therefore uniform deceleration of the control drum is impossible.

III. Linear Piston

The linear piston device is very much like the circular vane dashpot except that its motion is along a straight line. An example of a linear piston device using a linkage type coupling concept is shown in figure 5. The advantages of this device are basically the same as those of the rotating vane. In addition it appears that this device might be the simplest and easiest of all to fabricate. The main disadvantage of this device is that the linear action of the piston must be in some way coupled to the rotating shaft. Coupling methods using gears, levers, and cams were considered. Because of the high temperature and the liquid lithium environment these components must be made of compatible refractory material such as T-111. Under these conditions it is undesirable to have friction between moving parts of similar material. An alternative to this would be the use of cermet components. All of the coupling devices considered for use with the linear piston were rejected for one or more of the following reasons:

1. Friction between moving parts
2. High impact loads (also precludes the use of cermets)
3. Space allotment was exceeded

Of the three types of damping devices considered, the rotating vane type seems to have the fewest problem areas. The only disadvantage is the difficult machining required to control the clearances. The other two devices have problems which are serious enough to consider their feasibility doubtful.

Performance Analysis of Rotating Vane Dashpots

The analysis will be limited to rotating vane dashpots. The objectives of the analysis are to determine the braking characteristics that are possible using vane type dashpots with various values of clearance between the vane and the walls of the housing and to determine the clearances which are required to produce a uniform deceleration of the control drum in the reference design reactor.

A typical rectangular vane dashpot was shown in figure 2. The braking action of this dashpot occurs in the angular section indicated by θ. (See the list of symbols and definitions at the end of the report.) The braking action starts when the rotating vane reaches this section where the clearance is stepped down suddenly. The load caused by this sudden step is not severe and will not damage the drive train. The angular velocity at the start of braking is at its maximum, ($\omega = \omega_{max}$). The instantaneous velocity, V_v, at the center of the vane is:

$$V_v = R\omega$$

and at the start of braking is:

$$V_v = R\omega_{max}$$

As the vane rotates, fluid is forced to flow from the volume in front of the vane to the volume behind the vane. This fluid flows through the orifice formed by the gap or clearance between the moving vane and the housing. The instantaneous mass flow rate, W, through this orifice is:

$$W = \rho V_v \, Av$$

The Reynolds number, Re, was computed by the following equations:
(for circular vane models)

$$Re = \frac{W(D - d)}{A\mu}$$

(for rectangular vane models)

$$Re = \frac{2W}{\mu(Z + C)}$$

$$\text{where } Z = 2B + H$$

The ΔP, $(P_1 - P_2)$ was computed using equations from reference 3, which are basic orifice flow equations modified to apply to close-clearance orifices. When the flow clearances are only a few tenths of a millimeter or less, capillary effects take place and ordinary pressure drop equations are no longer accurate. The term K, (orifice coefficient), is an emperical coefficient which is used to correct the pressure drop equation.

The ΔP across the vane is computed by solving simultaneously the following equations (from ref. 3):
(for circular vane models)

$$W = \frac{\rho V \pi (D^2 - d^2)}{4}$$

$$V = K \sqrt{\frac{2\Delta P}{\rho}}$$

(for rectangular vane models)

$$W = \rho VZC$$

$$V = K \sqrt{\frac{2\Delta P}{\rho}}$$

Reference 3 presents charts correlating orifice coefficients (K) against the ratio C/L for families of constant Re. These data were compiled from numerous published test results. The values of K presented are mean values obtained from the published data. Empirical solutions based on the mean values of K agree with the published test results to within ± 15 percent.

Figure 6 shows the chart of K against C/L used in the example calculations.

The braking torque T_B can be computed from the equation:

$$T_B = \Delta P A_V R$$

The net braking torque, T_{BN}, can be computed from the equation:

$$T_{BN} = T_B - T_S$$

where T_S is the available torque of the scram spring (total scram spring torque minus the losses caused by friction in the drive train).

T_{BN} is the braking torque that is available for counteracting the inertia of the rotating control drum. The angular deceleration, α, is therefore:

$$\alpha = \frac{T_{BN}}{I}$$

where I is the moment of inertia of the control drum.

The change in angular velocity, $\Delta\omega$ is computed by assuming a small braking period, Δt.

$$\Delta\omega = \alpha \Delta t$$

The instantaneous angular velocity at the end of the first period, Δt, is:

$$\omega = \omega_{max} - \Delta\omega$$

This procedure is repeated for additional braking periods until either the dashpot reaches the stop or until the net braking torque becomes negligible. One method of determining when the dashpot reaches the stop is to plot ω against time. The area under the curve is the angular distance traveled by the rotating vane during braking.

$$\text{Angular Distance} = \sum_{1}^{n} \bar{\omega}_n \Delta t_n$$

Figure 7 shows the three models used in the analyses. Models 7a and 7b were selected to compare the braking characteristics of a rectangular and a circular vane design both of which have approximately the same external dimensions. These external dimensions are close to the maximum space alloted in the reference reactor design concept. Model 7c was analyzed to observe the braking performance of a dashpot of smaller size. Using these models, a range of constant clearances was selected and the analysis was done as described.

The following conditions were used for all of the three models: angular velocity at the start of braking = 15.7 radians per second, moment of inertia of the control drum = 0.298 Kg Meters^{2*}, available scram spring torque = 17.6 newton meters^{2}, (the value of torque required to accelerate the control drum from 0 to 15.7 radians per second in 0.267 sec.), lithium temperature = 1220 K. The following values of clearance were used: 0.0254 millimeter, 0.0508 millimeter, 0.0762 millimeter, 0.1270 millimeter, 0.1778 millimeter, and 0.2540 millimeter.

In obtaining orifice coefficients from chart 1 of reference 3, a C/L ratio of 1.0 was assumed. This assumption is equivalent to having vane edges which are equal in thickness to the clearance. This was done to work on the flat portion of the curves (fig. 6) where a slight variation in C/L has very little affect on orifice coefficient. This assumption is realistic if the vane edges are made very thin (on the order of a few hundredths of a millimeter). Thicker vane edges will produce better braking performance but the value of the orifice coefficient is difficult to determine for small values of C/L due to the extreme slope of the curves in that region of chart 1, reference 3 (fig. 6). The effect of this on the analysis is that the braking performances predicted are the minimum possible.

Figures 8 to 10 show the angular velocity against time after start of braking predicted by the analysis for assumed values of clearances for each of the three models analyzed. As expected the deceleration of the control drum is greater as the clearances become smaller. Typically for all values of clearances the angular velocity starts at the maximum value at the beginning of braking and diminishes with time until reaching a point where the velocity flattens out and becomes constant. At this point the braking torque equals the available scram spring torque and the net braking torque becomes zero. The control drum therefore continues the rest of the way to the stop at a constant velocity. In general, the smaller the clearance, the lower the final velocity becomes.

Figure 11 again shows the braking characteristics of the rectangular vane configuration along with the desired uniform deceleration curve. It can be seen that a constant clearance design cannot produce the desired deceleration. For the device to work properly, the clearance must be varied along the path of travel of the vane. There are several methods of determining the clearances required. One of the methods is to assume a constant net braking torque, T_{BN}, and a set of clearances and solve the equations of the

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*The control drum was assumed to have a mass of 101 kilograms and a radius of gyration of 5.43 centimeters.

analysis for angular velocity. This requires a trial and error selection of orifice coefficient, K, to find the proper Re. This method will give the clearance required for a given vane velocity. Another method is a graphical one and is the one which was used. For example, it can be seen in figure 11 that the deceleration obtained with a clearance of 0.245 millimeter is quite close to the desired deceleration curve up to about 0.04 seconds after the start of braking. The area under the curve up to this time is the angular distance traveled by the vane from the start of braking. At this point the clearance is reduced to the next smaller value shown on the graph (C = 0.178 mm). This is done by going to the curve for C = 0.178 millimeter at the same value of velocity as the final velocity at the end of the section having a clearance of 0.245 millimeter. This value of velocity is indicated by "a" on the 0.245 millimeter curve and by "a" on the 0.178 millimeter curve. The deceleration curve for the new clearance (0.178 mm) is used as long as it is close to the desired curve. This is done by obtaining values of $\Delta\omega$ against Δt (slope = α) starting at "a" on the 0.178 millimeter curve and translating them in time to a point starting at "a" on the curve for C = 0.254 millimeter. The slope for the 0.178 millimeter clearance is translated in time by increments until a point is reached where it is decided that the difference between the curve being generated and the desired deceleration curve has reached a tolerable limit. This velocity is indicated in this case by "b" on the curve being generated and by "b'" on the 0.178 millimeter curve. At this point it is again necessary to make another reduction in clearance to the next lower value shown on the graph (C = 0.127 mm). The value of velocity used at the start of this section is indicated by "b''" on the 0.127 millimeter curve. The slope is translated in the manner previously described. In this case the curve for C = 0.127 millimeter was considered within a tolerable limit of difference from the desired curve until the value of velocity was reached indicated by "C", "C'", and "C''" on the curves. Further reductions in clearance are made in the same way until the minimum allowable value of clearance is reached, indicated on the curves by "d" "d'", and "d''". For this case the minimum value is 0.0508 millimeter.

Figure 12 shows the clearances required along the angular path of the vane to produce uniform deceleration for the three models. The rectangular vane dashpot can produce the desired deceleration curve with the largest clearances of the three models. The smaller sized circular vane dashpot requires the smallest clearances of the three models. The rectangular vane design is therefore the least critical in the precision required for the clearances that are necessary to produce the desired deceleration of the control drum.

RESULTS AND DISCUSSION

These calculations were done for rather thin vane edges for the purpose of determining the value of orifice coefficient, K, with a greater degree of certainty. Use of thicker vane edges would result in more effective damping and will probably allow the clearances to increase with no sacrifice in braking performance. The clearances may be increased with the use of thicker vanes as long as flow conditions through the slit remain in the regime required for the curves in figure 6.

In an actual dashpot design, rather than attempting to control the clearance between the vane and the chamber walls at the entire periphery of the vane, two alternate methods were considered more practical. One method is to control the clearance along only one edge of the vane and the corresponding housing wall while the other corresponding clearances between wall and vane are kept at the minimum allowable value. The other method is to use a constant clearance along the entire periphery of the vane and housing with a variable cross sectional groove machined into one wall of the chamber to provide the required variable orifice.

Calculations were made to determine the torsional stress in the portion of the drive shaft between the dashpot and the control drum that would occur if the stop were struck with the final drum velocity predicted by the analysis for the rectangular vane dashpot with the clearances shown in figure 12. As mentioned earlier this value of final velocity is somewhat pessimistic due to the assumptions made with regard to the ratio, C/L. The calculated stress in the reference design drive shaft (3.18 cm diameter) using this value of velocity is 41.5×10^{3} newtons per square centimeter which is well below the maximum allowable value for a material like T-111 at 1220 K.

SUMMARY OF RESULTS

The following results summarize the dashpot analysis as applicable to the reference design of the space power reactor.

1. Vane type dashpots with properly controlled clearances (or orifices) can decelerate the control drum at the desired rate.

2. Of the vane shapes analyzed, the rectangular vane appears to be the easiest to fabricate.

3. Final velocities obtainable with vane type dashpots are low enough not to cause damage upon hitting the stop.

4. The size of a workable vane type dashpot is consistent with the space allotment in the reference reactor design concept.

SYMBOLS

A	orifice area, m^2
A_V	vane area, m^2
B	vane height, m
C	clearance, m
D	$d + 2C$, m
d	diameter of circular vane, m
H	vane width, m
I	moment of inertia of control drum, 0.298 $kg \cdot m^2$
K	orifice coefficient
L	vane edge thickness, m
P_1	Li pressure on front side of vane, N/m^2
P_2	Li pressure on back side of vane, N/m^2
ΔP	$P_1 - P_2$, N/m^2
R	distance from vane center to drive shaft centerline, m
T_B	braking torque, N/m
T_S	available scram spring torque, 17.6 N-m
T_{NB}	net braking torque, $T_B - T_S$, N-m
t	time after start of braking, sec
Δt	increment of braking time, sec
V	local velocity of fluid at orifice
V_v	vane center velocity, m/sec
W	mass flow rate, kg/sec
Z	total slot length, m
α	angular deceleration, $radians/sec^2$
θ	effective portion of the dashpot, 1.05 radians
μ	viscosity of Li at 1220 K, 2.06×10^{-4} $N\text{-}sec/m^2$
ρ	density of Li at 1220 K, 462 kg/m^3
ω	angular velocity, radians/sec
ω_{max}	maximum angular velocity, 15.7 radians/sec

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2. Krasner, Morton H.; Davison, Harry W.; and Diaguila, Anthony J.: Conceptual Design of a Compact Fast Reactor for Space Power. Presented at the Amer. Nucl. Soc. Annual Meeting, Boston, Mass, June 13-17, 1971.

3. Lenkei, Andrew: Close-Clearance Orifices. Product Eng., vol. 36, no. 9, Apr. 26, 1956, pp. 57-61.

COMPACT FAST REACTOR-REFERENCE DESIGN

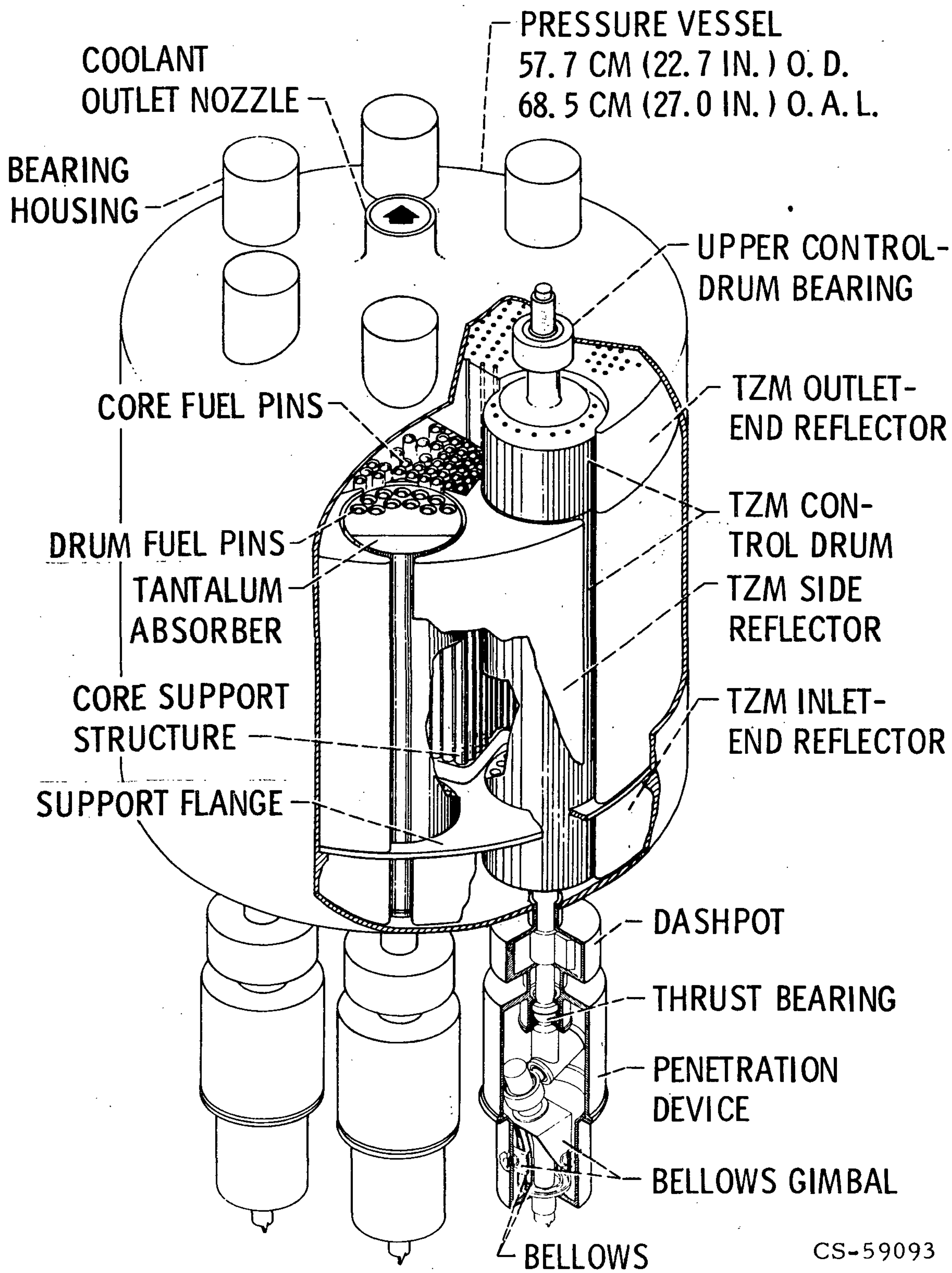


Figure 1

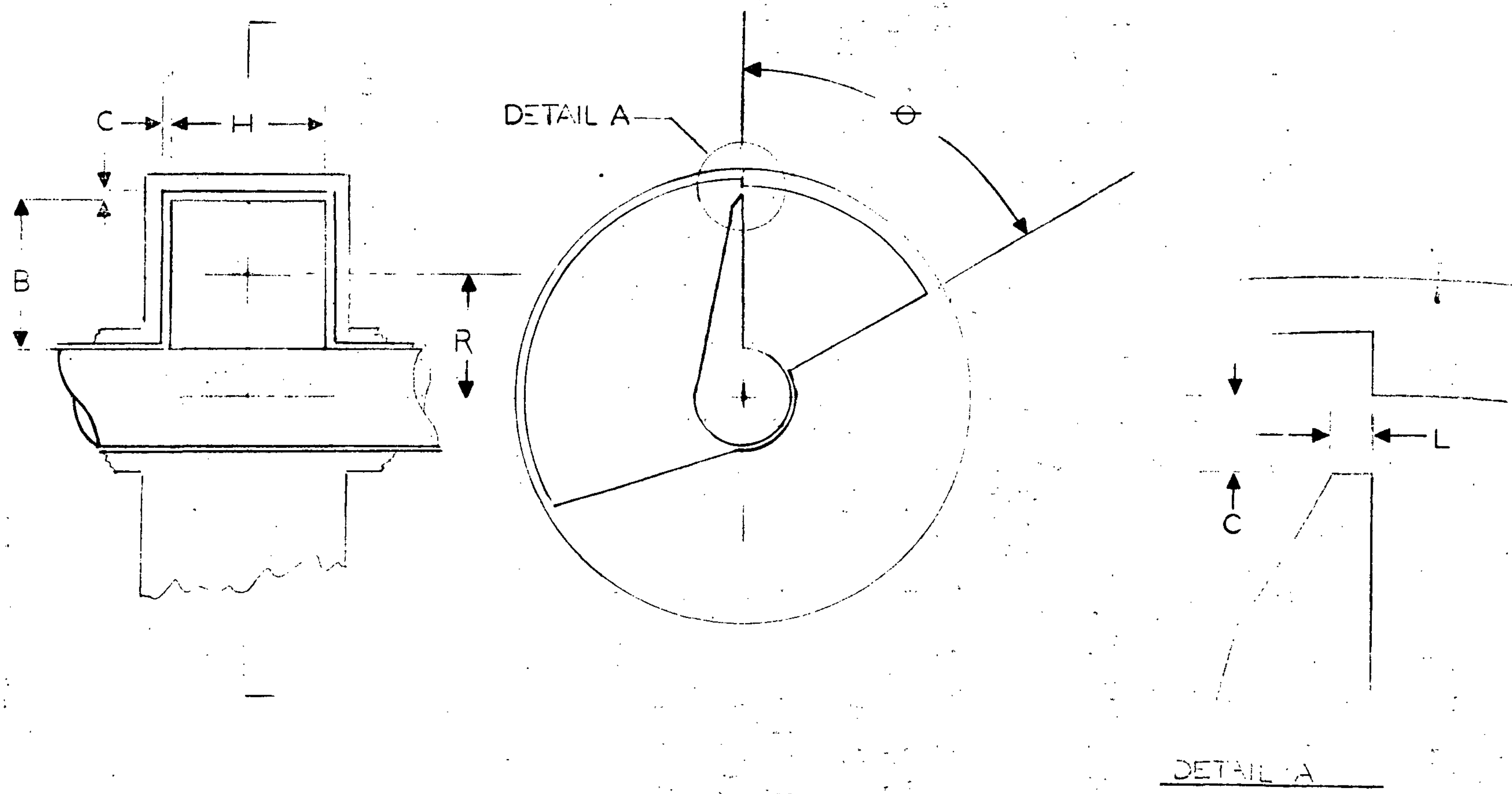


Figure 2 - Rectangular vane dashpot

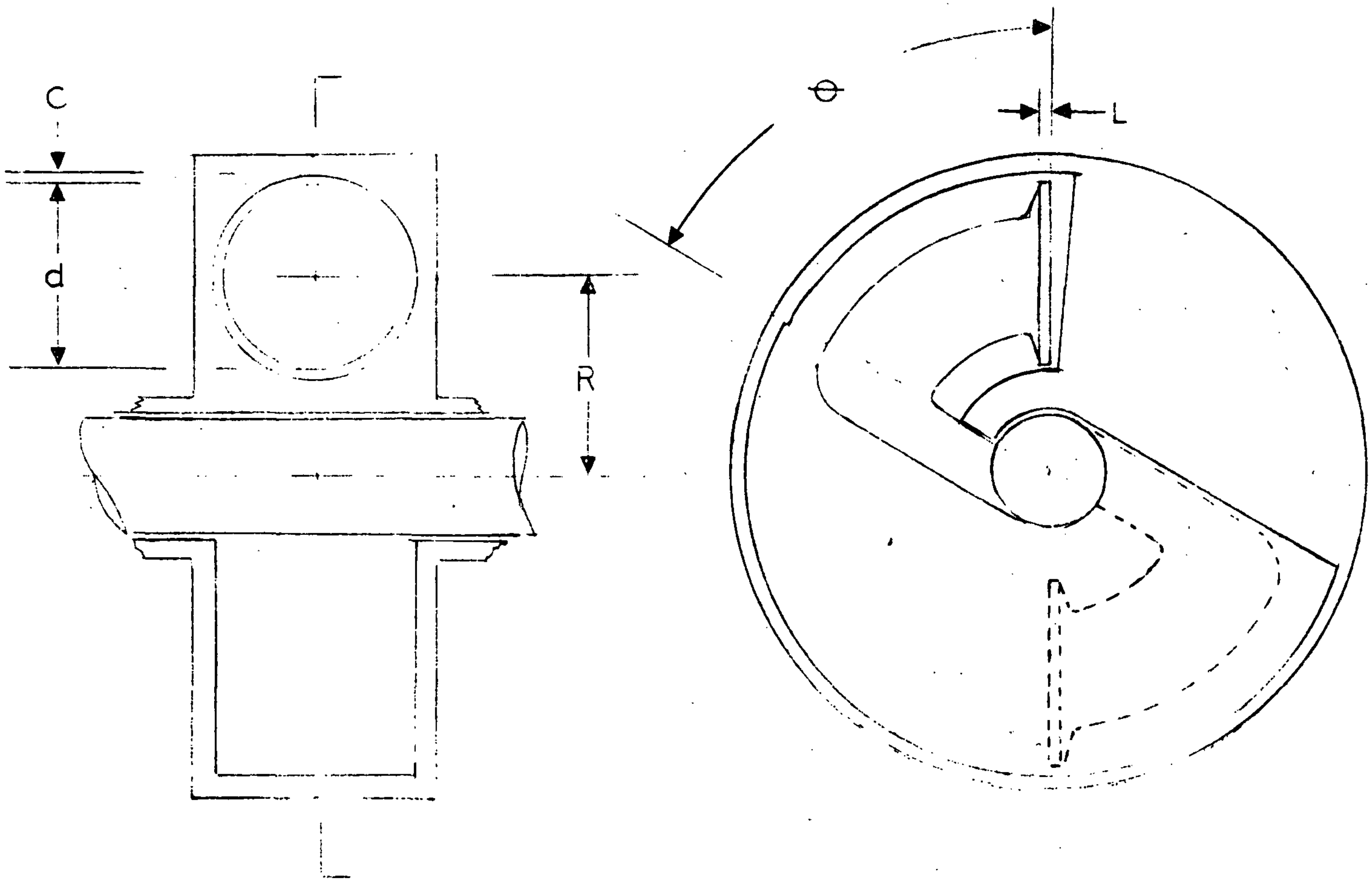


Figure 3 – Circular vane disrupt

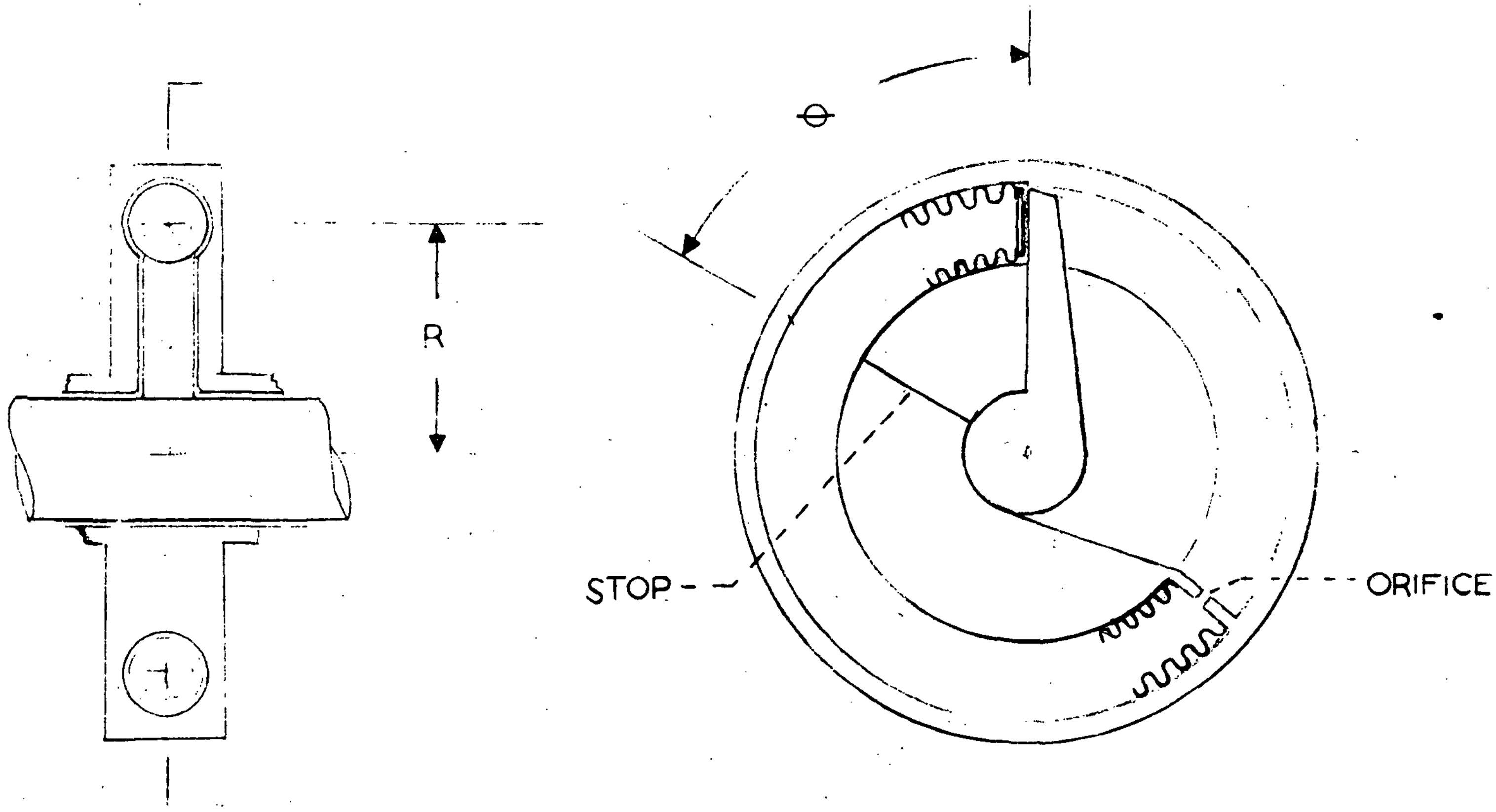


Figure 4 - Bellows type dashpot

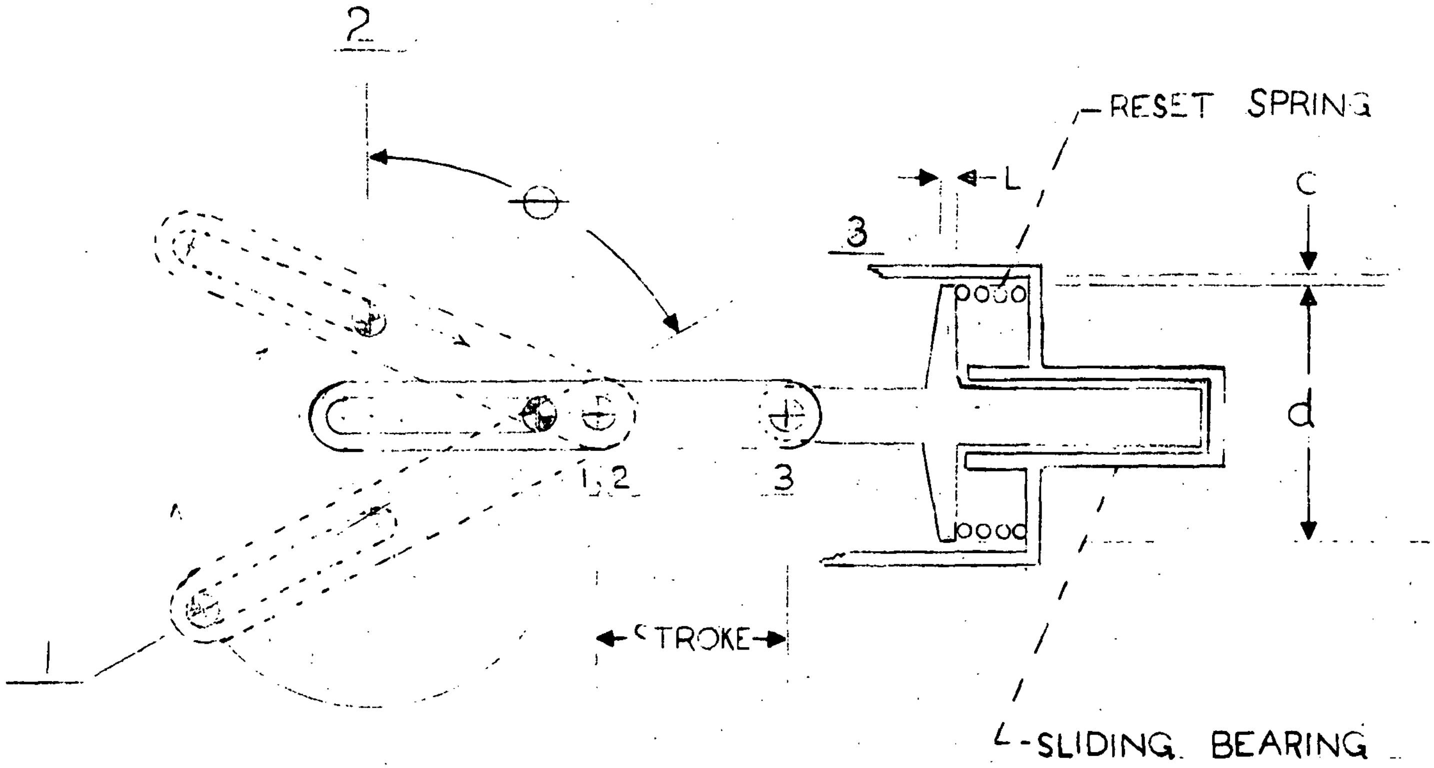
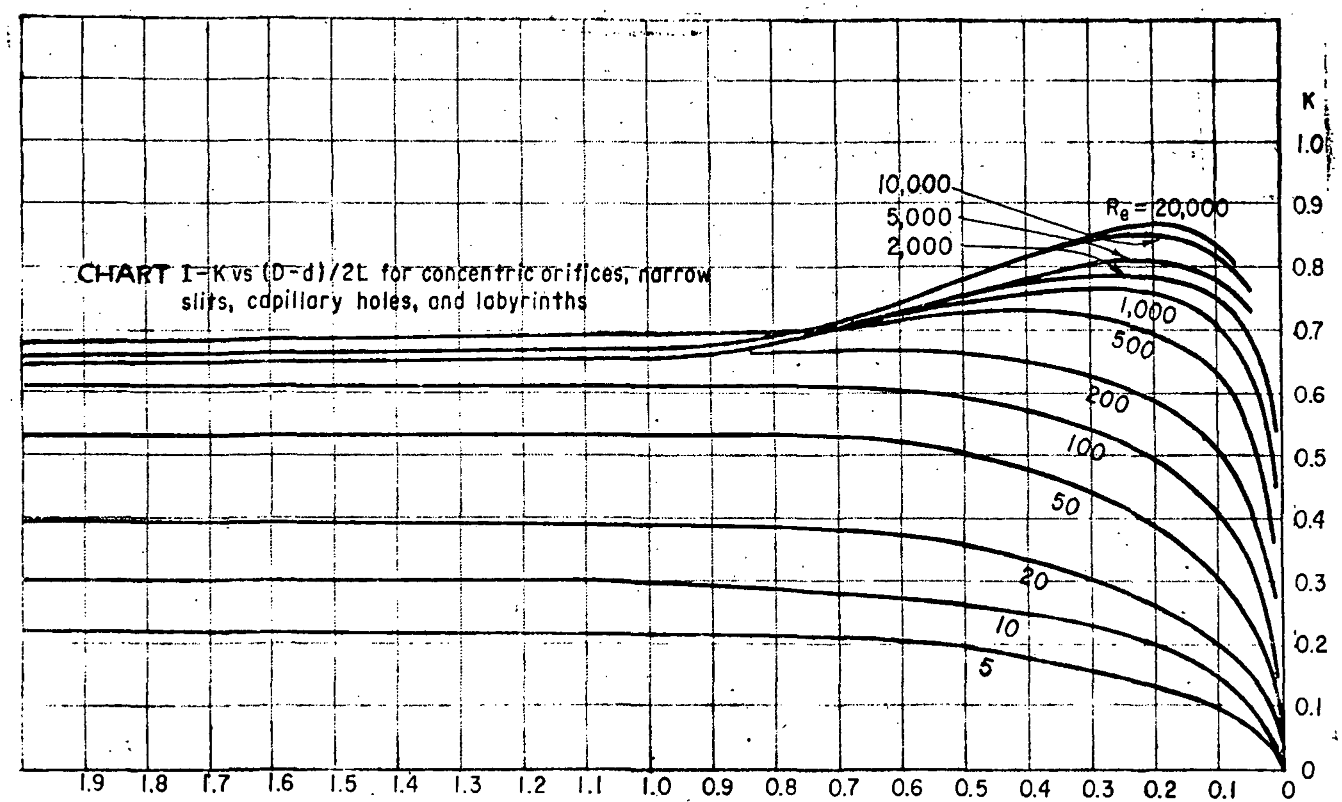
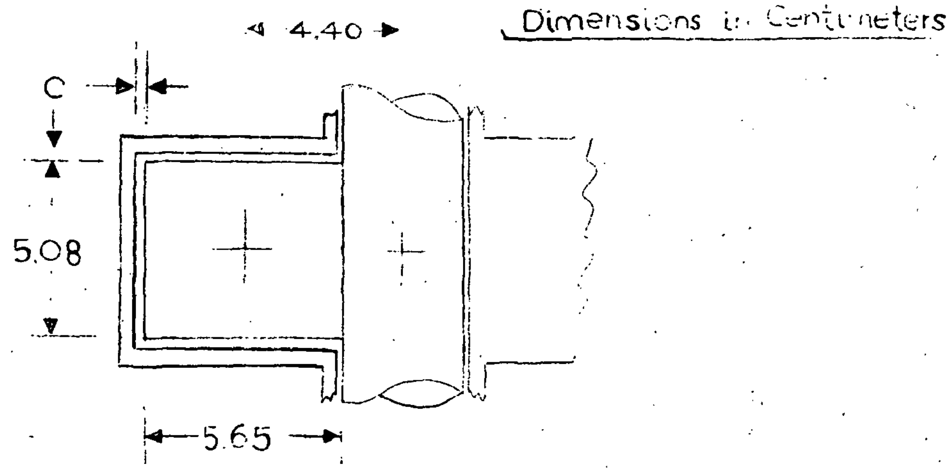


Figure 5 - Linear piston type dashpot

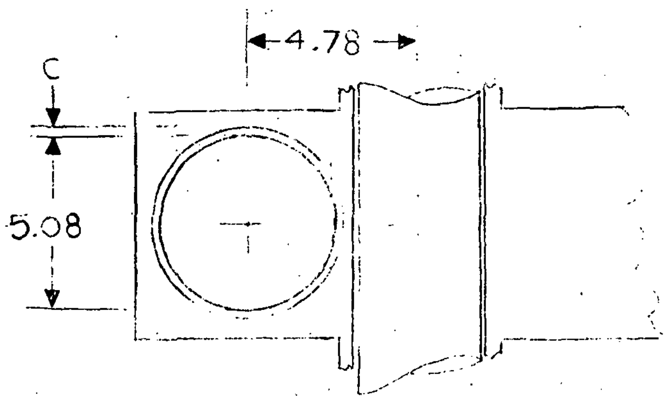
Figure 6 -Orifice Coefficient , K , vs. C/L from ref. 3



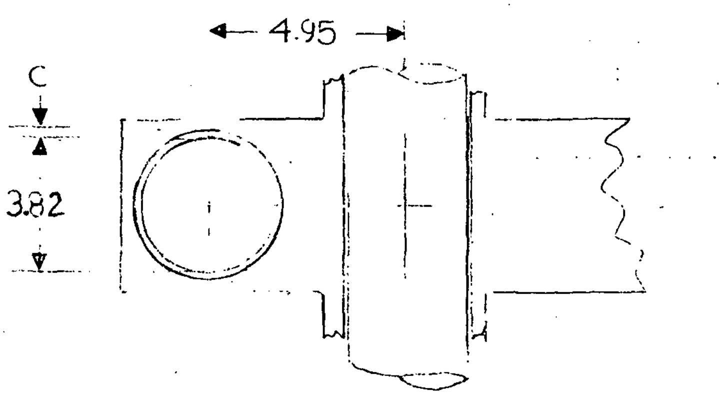
C/L , (CLEARANCE)/(FL W LENGTH)



a. RECTANGULAR VANE MODEL



b. CIRCULAR VANE MODEL



c. SMALL CIRCULAR VANE MODEL

Figure 7 - Analytical Models

Figure 8 - Braking characteristics of the rectangular vane dashpot of figure 7-a

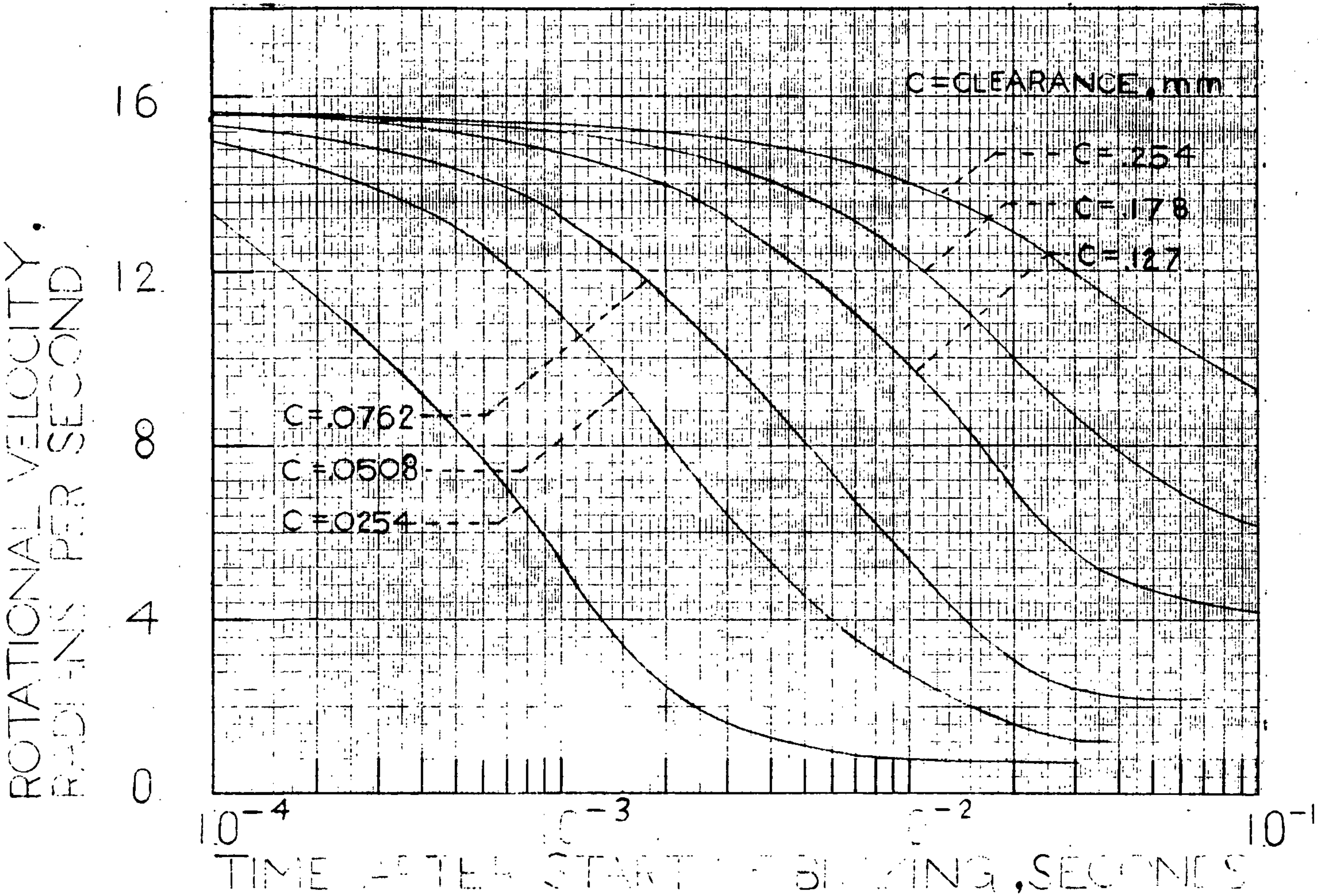


Figure 9-Braking characteristics of the circular-vane dashpot of figure 7-b

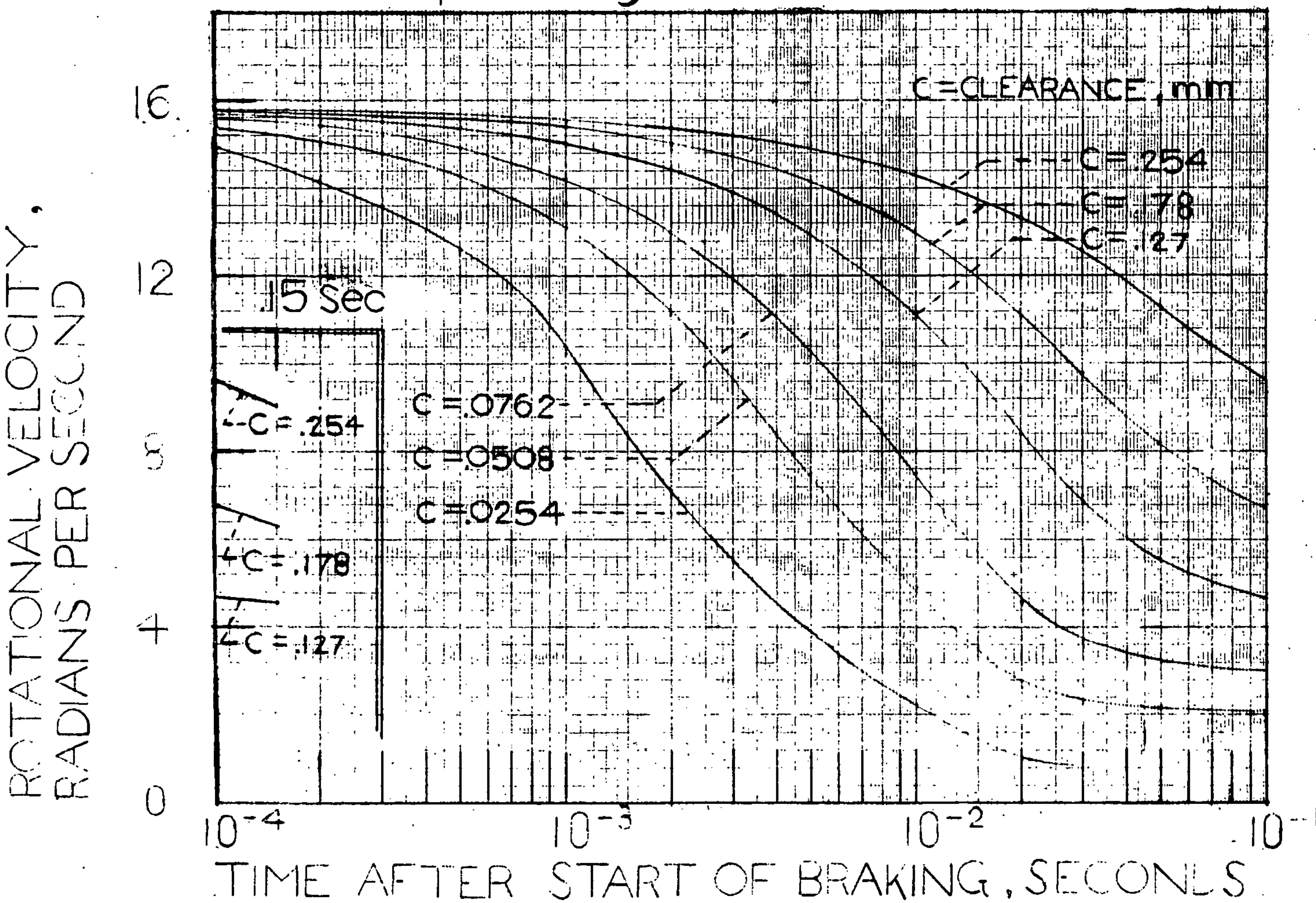


Figure 10 – Braking characteristics of the circular vane dashpot of figure 7-c

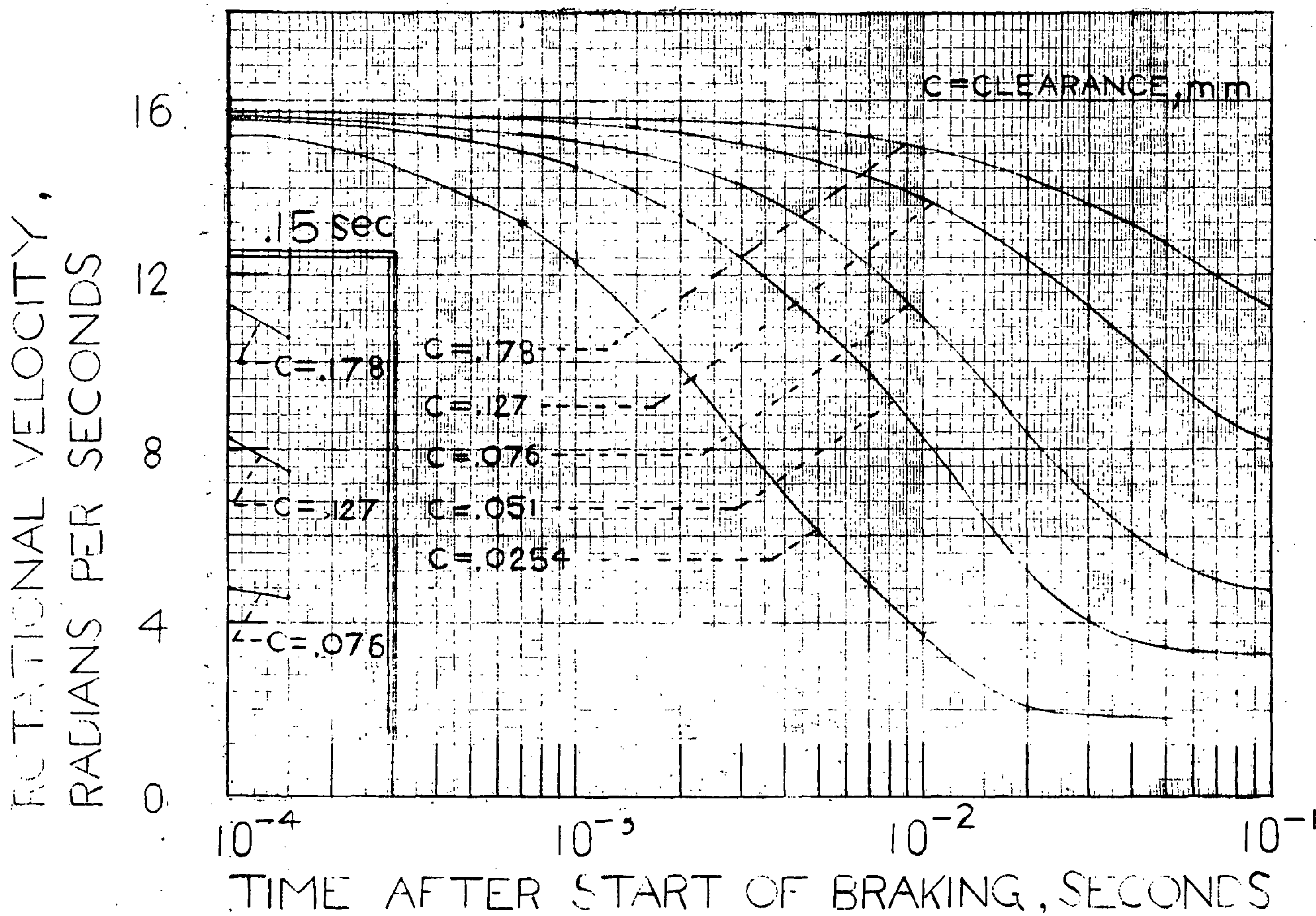


Figure 11 - Illustration of the graphical method of determining the required clearances for the dashpot of figure 7-a

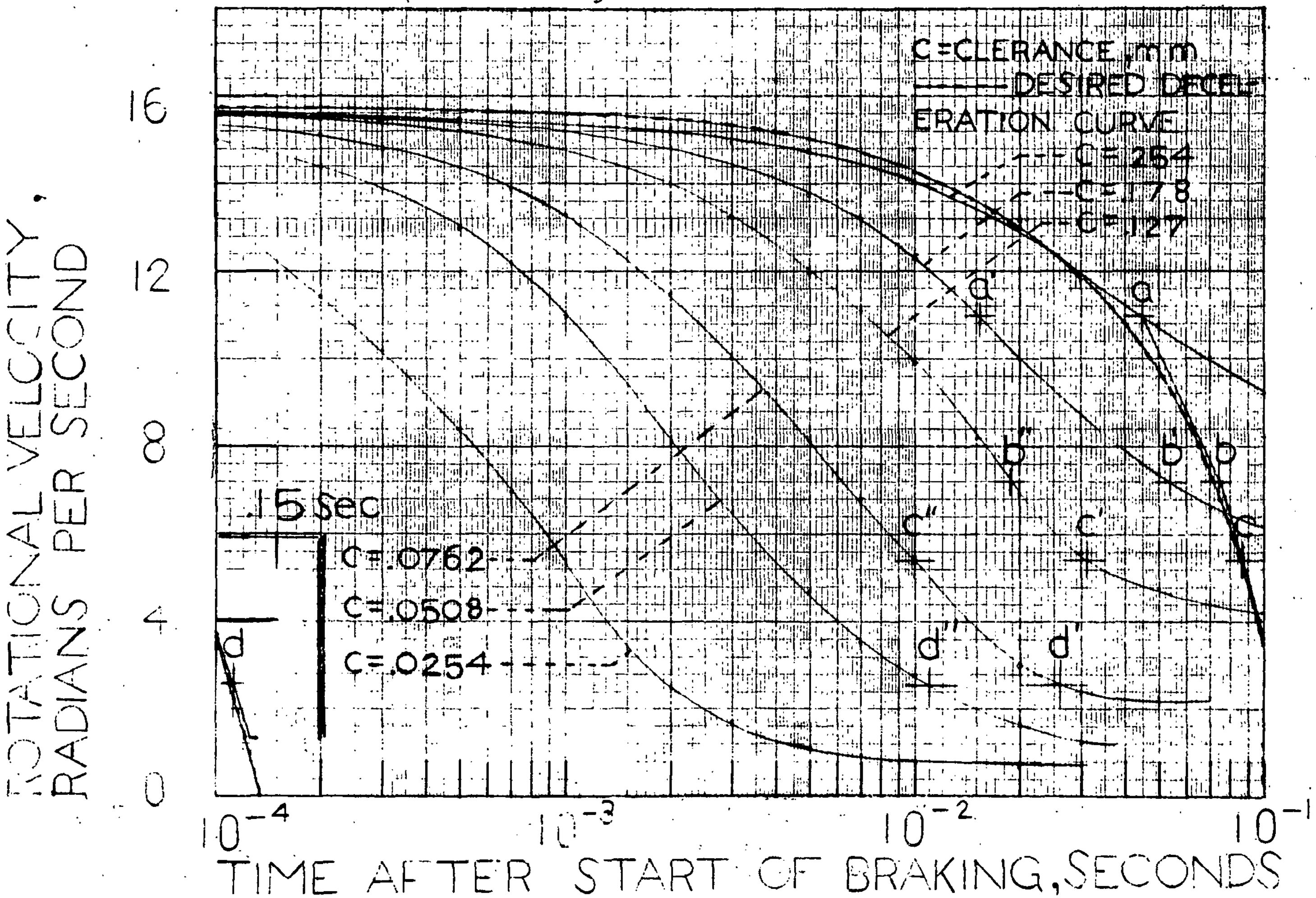


Figure 12- Clearance vs. angular position for the dashpots shown in figures 6-a, b, and c